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THESIS

Fundamental Vibrational Frequency
by a
3D - FEM Procedure

by

Howard Lawrence Crego

June 1975

Thesis Advisor:

J. E. Brock

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Fundamental Vibrational Frequency
by a
3D - FEM Procedure

by

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Lieutenant, United States Navy
B.S.M.E., Purdue University, 1968

Submitted in partial fulfillment of the
requirements for the degrees of

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ABSTRACT

This thesis presents a computer program which will solve for the lowest natural vibrational frequency of an arbitrarily shaped three dimensional object. The program was developed using a finite element numerical model which may employ either linear or quadratic isoparametric elements. Necessary instructions and guides for the user are included.

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I. INTRODUCTION

This thesis presents and discusses a computer program, called NAT1-3D, which employs the finite element method to set up and solve the elastodynamic equations for the free vibration of a three-dimensional, linearly elastic body so as to emerge with an accurate approximation of the fundamental frequency of free harmonic vibration.

The investigation was originally motivated by the interesting, important, and unsolved problem of the effect of foundation compliance and foundation inertia upon the fundamental natural frequency of a short elastic cantilever beam. It is clear that the usual assumption that the root of such a beam is rigidly fixed does not adequately represent the fact that no foundation is truly rigid. Calculations based upon ideal root fixity thus necessarily result in estimates of the fundamental vibration frequency that are too high. There seems to be no reasonably reliable way of estimating this effect [Ref.1].¹

During the course of the work involved in adapting existing computer software to the problem indicated in the last paragraph (in particular adding a dynamic capability which such software did not previously possess) it became more and more apparent that it would be most useful to focus attention upon the development of a general purpose

¹References will usually be cited in this manner; see page 67 for list.

. program for the evaluation of the fundamental frequency of three dimensional elastic bodies of any shape. Because of the usual and not unexpected problems of testing and debugging a very complicated digital computer program, time was not available for the study of the root fixity problem. Furthermore, and more importantly, for this problem or any problem in which truly complicated geometry is involved, digital computer programs which rely solely on in-core storage, as does the program developed herein as well as most of its predecessors, are capable of supplying results of marginal accuracy so that parametric studies or studies involving systematic changes in geometry are apt to have limited success. This is because in-core storage limitations do not permit subdividing the body into sufficiently small parts to permit obtaining really accurate results. Accordingly, one of the recommendations made herein is that attention be focussed on modifying this program so as to permit efficient out-of-core storage and manipulation. The present study does demonstrate that the procedural sequence adopted and implemented herein leads to reliable evaluations if the geometry of the vibrating object is sufficiently simple that it may be represented by a reasonably small number of finite elements.

Although the immediately following sections (II, III, IV, V, and VI) discuss theory, sources, and details of the program developed herein, the principal thrust and value of this

thesis is felt to be the detailed instruction provided for the employment of the program NAT1-3D by a new user for the actual evaluation of the fundamental frequency of a three-dimensional elastic body in which he may be interested.

II. THE FINITE ELEMENT METHOD

In recent years a procedure, called the Finite Element Method, has been developed and widely used for a variety of engineering problems involving the analysis of bodies having complicated shapes. In the analysis of the stresses and deformations of an elastic body, using this method (which is generally abbreviated as FEM), the body is subdivided into a (large) number of geometrical subdivisions called finite elements. The adjoining surfaces of these elements are delineated by points called nodes. In a three-dimensional problem each node may have three displacements. Displacements, and, by a process of differentiation, strains at points interior to such a finite element are expressible in terms of nodal displacements by use of so-called shape functions which, roughly speaking, are polynomials in the nodal displacements. The laws of elasticity are used to relate nodal displacements of a finite element to the forces applied at its nodes and the relationship is conveniently expressed in the form of an element stiffness matrix.

An assembly algorithm is used to combine such element stiffness matrices into a so-called global stiffness matrix, for the original entire body. This global stiffness matrix, hereinafter designated as [BGK], relates nodal displacements for the entire body to nodal forces which may be applied to

it. If there are a large number of finite elements in the body and if each of them has itself a substantial number of nodes, the total number of displacements, three for each node, may be very large.

In elastostatic problems, the nodal forces, i.e., the force components applied to the nodes, are arranged in the form of a vector and the unknown nodal displacement components are similarly arranged, so that the relationship can be expressed in the matrix equation

$$[BGK] \{v\} = \{f\}$$

If the size of the vectors $\{v\}$ and $\{f\}$ were not great, any of a number of procedures could be used to solve this matrix equation for the unknown vector $\{v\}$. However, in practice, the size usually is quite large, and severe problems of storage in an electronic computer have resulted in the development of quite subtle procedures for storing the elements of $[BGK]$ and of $\{f\}$.

The most significant features of the matrix $[BGK]$ are that it is positive definite, symmetric, and of banded structure. The last means that there is an upper right and a lower left triangle which consists entirely of zero elements which need not actually be stored if their presence is otherwise suitably noted. These properties permit storing the $[BGK]$ matrix in far less space than if it were to be stored in the form usually used to exhibit square matrices.

Principle features of any FEM computer program are the generation and compact storage of the elements of [BGK], and a so-called "equation solver" which efficiently solves the linear algebraic problem indicated by the equation above. Software to accomplish this has been developed and tested and is readily available. One of the problems facing a person who undertakes to solve an engineering problem by FEM is to select among several competing software programs. TRISOP [Ref. 3] and FIELD [Ref. 4] are the names of programs developed by Professor G. Cantin, LCDR E.A. Leonidas and LCDR G.T. Lew. These programs have been successfully employed at NPS for a variety of investigations. It was decided to adapt and employ appropriate sections of these programs for the purposes of this thesis.

TRISOP and similar programs require a massive input to describe the geometry of the body and the manner in which it is subdivided, to introduce the physical properties of the materials, to represent the manner in which the body is constrained, and to account for the applied loadings. For example, in a typical problem treated in the course of this thesis investigation 250 data cards would be required as input. Since the labor of generating the data for and actually punching these cards is so great and since the likelihood of making at least one crippling error is almost certain, software programs called "mesh generators" have been developed which accept a minimum of input data and

instructions and which produce data cards suitable for subsequent use in an FEM program. Such a mesh generator, called TRIMEG [Ref. 2], developed by LT J.R. Adamek, for use with TRISOP has been used and tested at NPS, and has provided a very useful, almost essential adjunct to the FEM program used for this thesis.

TRISOP does not have a dynamic capability. As will be indicated in the next section hereof, one of the important features of a dynamics FEM program for use in determining natural frequencies is an appropriate mass matrix. Without going into detail, it may be stated that the theory of the FEM provides for the construction of so-called consistent mass matrices for the individual finite elements and their assembly into a global mass matrix [BGM]. The computational details are almost exactly the same as for the construction of element stiffness matrices and their assembly into [BGK]. The global mass matrix has essentially the same properties as the global stiffness matrix and requires the same storage.

Much thought was given to the problems of storing such a consistent mass matrix. However, studies by others [Ref. 7] have indicated that when evaluating the fundamental harmonic frequency of free vibrations of an elastic object, use of a lumped-mass form of the mass matrix gives results as good as or better than those obtainable by using a consistent mass matrix. The reasons for this are not clear,

but the advantages are. A lumped-mass mass matrix is diagonal in form and can be stored as a vector; briefly, using the lumped-mass form results in a dynamics problem requiring very little more storage than an elastostatic problem requires.

Accordingly it was decided to employ the lumped-mass form and the question arose of how optimally to obtain such a mass matrix. The procedure which finally evolved was to construct the element consistent mass matrix and to diagonalize it before assembly into the global mass matrix.

The word diagonalize in this context is not intended to mean the mathematical process of diagonalization by means of solving the complete eigenvalue problem for the matrix M . Instead, it merely means replacing the consistent (element) mass matrix by a diagonal matrix corresponding to an appropriate division of the mass of the element into submasses concentrated at the nodal points.

The assembly was done in the core storage space reserved for assembling (and later operating upon) the global stiffness matrix, and the assembly algorithm is exactly the same for both matrices. The assembled mass matrix is diagonal and its nontrivial elements are quickly stored elsewhere in the form of a vector.

The details of diagonalizing the element mass matrix deserve a brief description, explanation, and plausible justification. Aside from the fact that it is obviously

desirable to employ a procedure which involves a simple algorithm and results in the same diagonalized element mass matrix regardless of renumbering of nodes, a reasonable and plausible distribution of the total element mass into mass points at the nodes is also required. This matter is discussed in somewhat greater detail on page 29.

Accordingly appropriate segments of the programs TRISOP and FIELD were selected and modified so as to incorporate the construction of a lumped-mass mass matrix as outlined in the preceding paragraphs and so as also to incorporate the operations of LDLT decomposition, matrix (Stodola) iteration, convergence testing, and Rayleigh quotient evaluation discussed in the following section hereof. The resulting program is called NAT1-3D.

III. METHOD OF SOLUTION

With the use of the $[K]$ and $[M]$ matrices provided by the finite element formation, as discussed previously, the free vibration problem can be formulated as

$$[M] \{\ddot{v}\} + [K] \{v\} = \{0\} \quad (1)$$

The remarks which follow immediately represent standard procedure in vibration theory and are intended merely to recall to the reader the operational sequence which leads to the desired result. We are concerned with the lowest frequency harmonic vibrations which the system may execute. By assuming that the vector $\{v\}$ of time-varying displacements has the form

$$\{v\} = \{v^*\} \cos(\omega t)$$

(where $\{v^*\}$ is a vector of constant quantities representing the amplitudes of the motions of the nodal points) and substituting this equation into equation (1) above, the cosine term may be cancelled out, and, upon dispensing with the asterisk and henceforth regarding $\{v\}$ as a vector of constants, we arrive at the form

$$[K] \{v\} = \omega^2 [M] \{v\}$$

It may be shown that if an arbitrary vector $\{v_i\}$ is introduced on the right side and this equation is solved for a new vector, i.e., if we solve for $\{v_{i+1}\}$ in the equation

$$[K] \{v_{i+1}\} = \omega^2 [M] \{v_i\}$$

then $\{v_{i+1}\}$ is a better approximation to the theoretically correct vector for the lowest frequency than was $\{v_i\}$. Furthermore, if the vectors are suitably and similarly normalized, the scalar constant of normalization approaches more and more closely to the correct value of the square of the lowest frequency. This process of matrix iteration is frequently called Stodola-iteration after the Swiss engineer who introduced the method.

Although the scalar constant of normalization does provide an increasingly good evaluation of the desired value of ω_1^2 , at any stage of the iteration a more accurate estimate may be obtained from the Rayleigh quotient

$$\omega_1^2 \approx Q = \{v\}^T [K] \{v\} / \{v\}^T [M] \{v\}.$$

Performing this operation after a reasonable number of iterations provides the best currently available value for ω_1^2 and, presuming that the value so obtained is "close" to the values previously obtained during the Stodola iteration, provides a check on the accuracy and the validity of the entire computational process.

With systems having only a few degrees of freedom it is customary to solve for $\{v_{i+1}\}$ by obtaining the inverse of the matrix $[K]$. However, the desirable property of the $[K]$ matrix as obtained by the finite element method, namely its banded character which permits compact storage, is not possessed by the inverse $[K]^{-1}$; furthermore the operation of obtaining an inverse is computationally extravagant. Accordingly, it is very advantageous to represent $[K]$ in the so-called LDLT form, i.e., to decompose $[K]$ into the matrix product

$$[K] = [L] [D] [L]^T$$

where $[L]$ is a lower triangular matrix having unit diagonal elements (which need not be stored) and $[D]$ is a diagonal matrix. This decomposition is a standard operation which is not difficult to perform and the matrices $[L]$ and $[D]$ may be stored in the space occupied by $[K]$.

The normalization which is used is the so-called Euclidean normalization for which the unnormalized vector $\{v\}$ is divided throughout by the square root of the sum of the square of its elements.

The iteration proceeds as follows. Having the normalized vector $\{v_i\}$ (either as a result of having made a previous iteration or, the first time through, as the result of a guess or enlightened estimate) the vector

$$\{g\} = [M] [v_i]$$

is obtained simply by matrix multiplication. Then the vector $\{h\}$ is determined from the equation

$$[K] \{h\} = \{g\}$$

by a forward and backward substitution process making use of the desirable properties of the LDLT decomposition. Then $\{h\}$ is normalized so as to produce the next iterant $\{v_{i+1}\}$, the reciprocal of the constant of normalization being an approximation to ω_1^2 . This process is repeated until convergence is satisfactory. In the computer implementation of this procedure, described later, convergence was assumed to be satisfactory when two successive such evaluations of ω_1^2 differed by less than 10^{-6} . At this point, the Rayleigh quotient was evaluated in the form

$$Q = ([L]^T \{v\})^T [D] [L]^T \{v\} / \{v\}^T [M] \{v\}$$

Although the procedure described above could be employed with nondiagonal $[M]$, the fact that it was decided to diagonalize $[M]$ in order to minimize core storage requirements also made the computations involving $[M]$ very simple and economical.

IV. PROGRAM ARRANGEMENT

To facilitate the adaptation of existing mesh generating schemes and to incorporate the strong points of existing software the general arrangement of the program is modular in nature. This also allows future changes to be accomplished more easily if desired.

The actual solution of a problem requires two basic steps: mesh generation and frequency evaluation.

A. MESH GENERATION

Mesh generation is the process in which the object to be analyzed is divided into an assemblage of small elements. It is a laborious and tedious task that should not be rushed into. Intuition guided by an intelligent study of the geometry of the object is often the best approach to use in choosing a mesh.

TRIMEG (Ref. 1, which is discussed in greater detail in Appendix C) is a computer program that performs mesh generation for three dimensional objects. The program automatically discretizes a given geometry and upon command will punch data decks containing nodal coordinate and element connectivity information. The solution routine presented in this thesis was written to accept data in the format of TRIMEG's output. The use of TRIMEG is not a necessity and any source may be used for the input data. However, the use of

TRIMEG to generate data cards for use with NAT1-3D is strongly recommended to save effort and to reduce the likelihood of error. The reader is advised to study and understand TRIMEG; Ref. 1.

B. FREQUENCY EVALUATION

1. Common Statements

Throughout the development of this routine a prime objective was to make it as simple as possible to use. Simplicity can however limit flexibility. To enable the user to avoid requesting excessive core space by adjusting the core requirements to meet the needs of his particular problem, the large data blocks are placed in easily changed COMMON storage areas. This adjustment allows the use of more nodes for problems with smaller band widths.

There are four labeled COMMON statements which may be used to modify core storage requested by NAT1-3D; this is done to avoid requesting more core than necessary and thus results in obtaining optimum job priority. The locations of the cards and size of the arrays are indicated in the listing in Appendix B. The following is a tabulation of the statement names and the subroutines in which they appear.

BIGS	MAIN, MERGE, QUAD, FORM, FASTEN, LTLD, SOLV
ENTER	INPUT, MERGE
ITER	MAIN, NORM
SOL	MAIN, SOLV

In addition to modifying the COMMON statements the dimension of the vector FMODE which appears in the main routine should be calculated and introduced in the appropriate DIMENSION statement.

2. Input

The required data consists of six basic card groupings. Their specific formats are listed in Appendix B.

a. TITLE (one card)

Any desired problem title may be used.

b. PROBLEM DATA (one card)

The card contains the basic problem parameters. Here the user chooses the type of shape functions that are desired and indicates the number of Gauss points he wishes to use in their integration.

c. MATERIAL PROPERTIES (one card per material)

These cards contain the elastic property data for the isotropic materials used in the object. The number used to represent a material is arbitrary but it must be the same numbering scheme used in the mesh generation.

d. NODE CARDS (one card per node)

These cards, which may be punched by TRIMEG, contain the node numbers and their global coordinates.

e. CONNECTIVITY MATRIX (two cards per element)

These cards, which may be punched by TRIMEG, contain the detailed arrangement of all the contributions of each element to the nodal displacements. As the name implies,

this matrix connects the elements and serves as the controlling agent which distributes the contribution of each element to the global stiffness and mass matrices.

f. BOUNDARY CONDITION CARDS (one card per boundary node)

The user may impose boundary conditions by restraining the motion of any node in any coordinate direction. The boundary condition cards indicate the node and the coordinate direction to be fixed.

V. EXAMPLE PROBLEMS

The method usually used to test computer programs is to solve a problem that has a known exact solution to serve as a comparison gauge for the computer solution. This presents a difficulty as the author is unaware of a truly three dimensional classic solution for natural frequency determination. Problems A and B, below may seem overly simple but they do have known approximate solutions and serve to establish a measure of confidence in the integrity of the routine.

The approximate solutions are based on Euler beam theory or on the more elaborate Timoshenko beam theory, both of which involve replacing the equations of three dimensional elasticity (theoretically required to describe the problem) by greatly simplified equations.

A. CANTILEVER BEAM

The cantilever beam is a basic design structure which may vary in form from the smallest man-made gear tooth to the largest trees produced by mother nature. The difficulty of analysis also may vary as drastically. The inclusion of effects such as shear and rotary inertia significantly complicates the frequency equation.

The specific structure used in this problem was a beam six inches long having a uniform one inch square cross section. Subdivision is indicated in Figure 1. The results were as follows:

Euler Beam Theory	$\omega = 5700$ rad/sec
NAT1-3D Results	$\omega = 5664$ rad/sec
Ratio NAT1-3D/Euler	0.994

This agrees with Timoshenko theory which predicts a frequency correction for shear and rotary inertia of nearly unity for this geometry [Ref. 8]. Figure 1 shows the convergence of the solution as the number of nodes is increased, i.e., the effect of finer subdivision. Note however, that the finer discretization involves only the "Z" direction. It would be of interest also to investigate finer subdivisions in the "X" and/or "Y" direction. However, this would involve substantial increases in bandwidth.

B. SIMPLY SUPPORTED BEAM

The effects of shear deflection and rotary inertia are more pronounced in a simply supported beam having the same geometry as the previous cantilever.

Euler Beam Theory	$\omega = 16,000$ rad/sec
NAT1-3D	$\omega = 14,250$ rad/sec
Timoshenko Beam Theory	$\omega = 14,000$ rad/sec

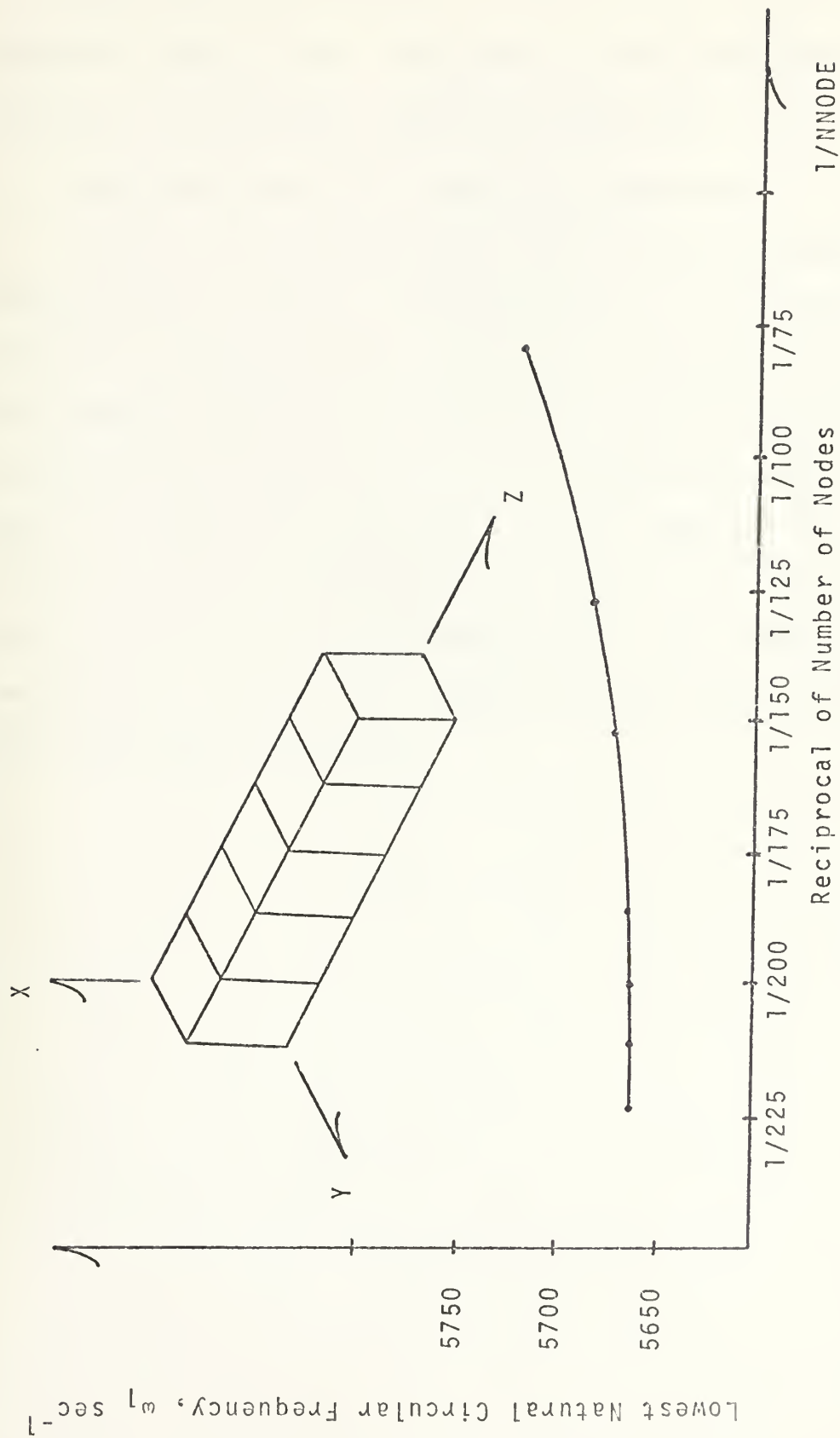


Figure 1. Convergence of Solution as Number of Nodes Increased.

The Timoshenko solution is based on an assumed shear constant value of 0.6. This value is the approximate middle of the accepted range of values [Ref. 8].

C. CANTILEVER BEAM WITH COMPLIANT FOUNDATION

Figure 2 represents a mesh which could be used to investigate the effects of root fixity on the natural frequency of a cantilever beam having imperfect root fixity. The boundary conditions would be applied to the outer wall surfaces thus allowing the root to deflect naturally. The material properties of the wall and the beam can also be varied to measure the effects of dissimilar materials. A final solution was not obtainable due to the large core requirements arising from even the rather coarse discretization indicated in Figure 2.

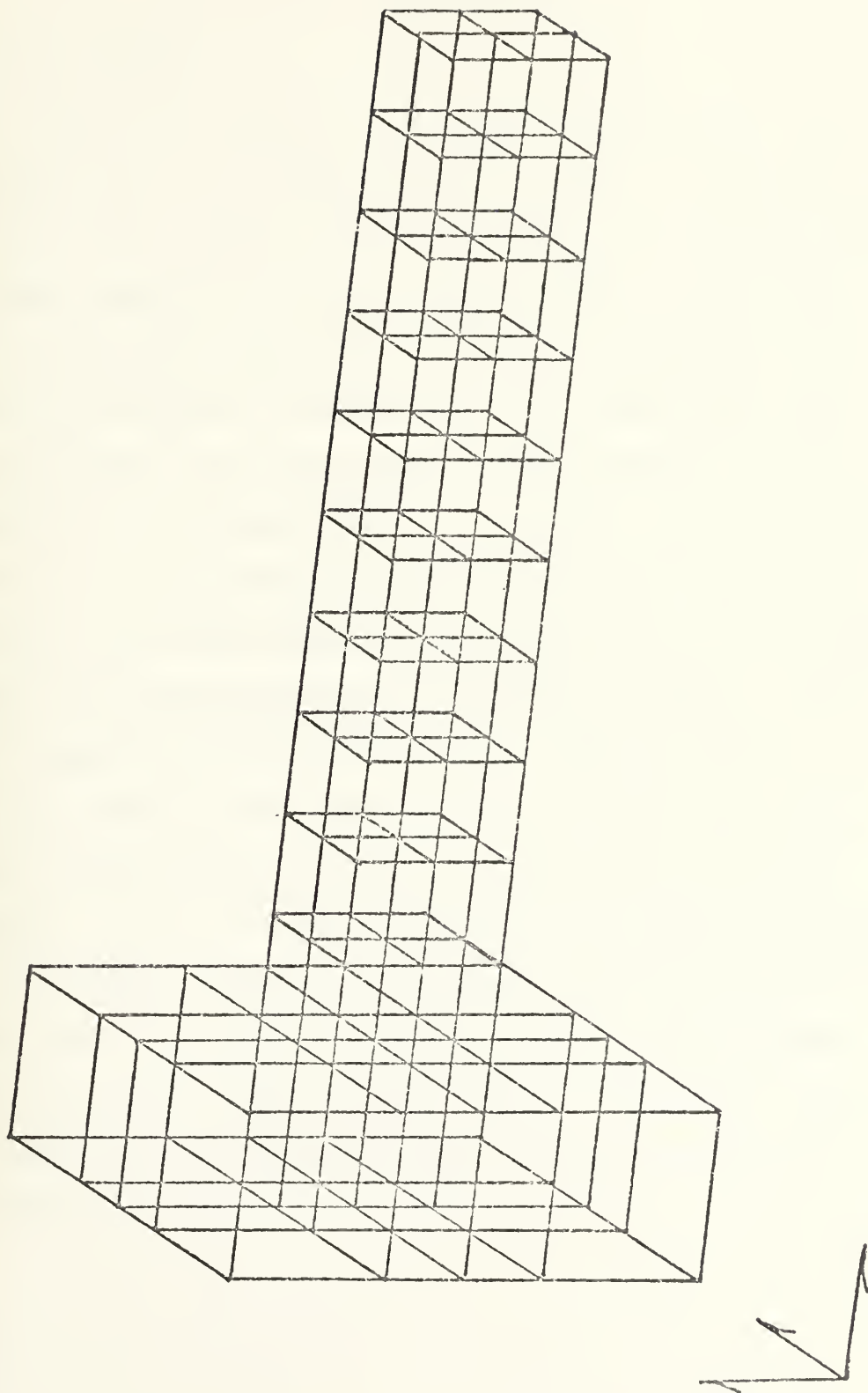


Figure 2. Discretization of Cantilever to Investigate Root Fixity Effects.

VI. DISCUSSION

A. MASS MATRIX

As noted in chapter II a lumped diagonal mass matrix was chosen for use in the solution routine. The choice was based upon the reduced core requirements and computational advantages inherent to the diagonal matrix. A decision was made to form the lumped matrix from the consistent matrix because the numerical evaluation of the consistent mass matrix parallels the evaluation of the stiffness matrix allowing the two formulations to be coded simultaneously. Lumping is accomplished by replacing the diagonal entry in each row of the consistent matrix with the sum of the entries in that row and zeroing the non-diagonal elements. The result could be termed a consistent, lumped matrix.

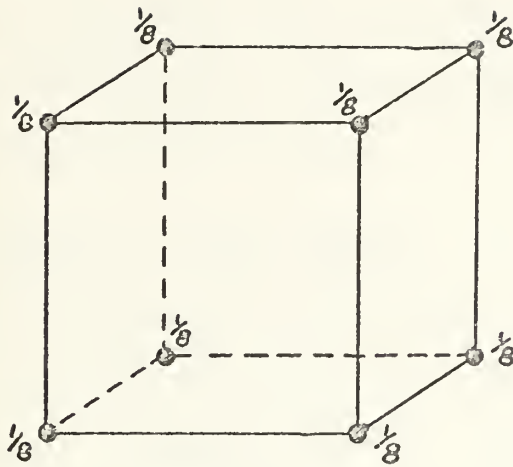
Reference 7, previously cited, indicate that this form of mass matrix yields excellent results for the lowest frequency. It is of some interest at this point to examine the point-mass distribution which results from this process in the case of an element in the form of a rectangular parallelepiped, a case for which physical intuition provides some indication of whether or not the substitution is reasonable.

Figure 3 shows the resultant mass distribution for both linear and quadratic elements. The distribution for the linear element appears logical. The quadratic element however is not at all obvious. The allocation of negative mass at the element corners would seem to conflict with common sense. These results are however, not without precedent. Negative corner values result when imposing uniform body forces on rectangular elements in a pattern that is nearly identical to that shown in Figure 1 of Ref. 6 (p. 168).

B. SHAPE FUNCTIONS USED

In its earlier versions NAT1-3D accommodated quadratic shape functions only. This was a compromise between the limitations of linear functions in representing complex geometry and the large core size requirements of cubic elements. A later inclusion of linear shape functions resulted in the observation that they were "too stiff." The answers they gave for the fundamental frequency were high when the same number of nodes and same nodal arrangement was used that corresponded to an accurate solution based on quadratic functions.

This is to be expected since, for the same subdivision into elements, the shape functions for quadratic elements permit a closer representation of the actual deformation pattern than is permitted by using linear elements. In effect this corresponds to both types of elements implicitly



Distribution of
Mass, Rectangular
Parallelepiped,
Linear Functions

Distribution of Mass,
Rectangular Parallelepiped,
Quadratic Functions

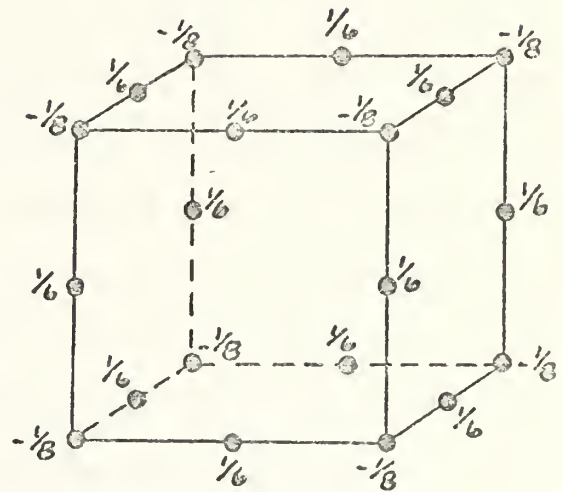


Figure 3. Allocation of Lumped-Mass Mass Matrix.

introducing fictitious and undesirable restraints, which increase the calculated frequency above its correct value; however, these restraints are greater and have a more serious effect in the case of linear elements than they do in the case of quadratic elements.

C. NUMBER OF GAUSS POINTS

If N represents the number of Gauss points, the Gaussian technique used in this routine will exactly integrate a polynomial of degree $(2N-1)$. It has been noted however, in calculations employing NAT1-3D that the minimum number of Gauss points which can successfully be used with quadratic elements is three. When fewer than three Gauss points are used it is found that accuracy of the frequency evaluation is seriously decreased. It is conjectured that this behavior is related to the method of introducing the boundary conditions (by multiplying selected diagonal elements of [BGK] by a large factor, "WELD"). The calculations which showed the undesirable behavior had $WELD = 10^{10}$. It would be interesting to look into this matter more fully by reducing the value of WELD, by considering alternate methods of introducing boundary fixity, and in other ways. Briefly, we do not presently have a satisfactory understanding of the effect of modifying the number of Gauss points except to observe that using $NGP = 2$ gave unsatisfactory results, while using $NGP > 3$ gave results which were felt to be no more accurate or reliable than those obtained by using $NGP = 3$.

D. CONVERGENCE

Our experience has shown that the Stodola iteration converges with amazing rapidity, three to four iterations usually being sufficient. This can be attributed to the fact that the structures considered had lower frequencies which were widely separated from the second frequency. It is expected that objects whose lower frequencies are not separated will require more iterations to converge.

It should be noted that the "initial guess," being a unit vector, contains elements of both transverse and axial vibrations. If the lowest mode were an axial mode it would be reflected in the corresponding elements of the stiffness matrix and the iteration should converge to the actual lowest frequency be it an axial or transverse mode. Correct solutions have been obtained for transverse vibration using a pure axial mode as an initial guess.

E. PROGRAM LIMITATIONS

It could be considered overkill to use this routine on a truly two dimensional plane strain or plane stress problem. During the development of this routine a 2-D version was written but it required exactly the same work on the user's part in the form of data preparation and further required the user to recognize that his object could be modeled as a plane stress/strain case. Upon considering that an original

goal was to limit the assumptions and idealizations that the user has to make, only the three dimensional routine is being presented.

In its present format the program is limited by core size. This problem arises when dealing with complex geometries which require several layers of elements in more than one direction. Multidirectional layering results in a dramatically rapid growth in the band width of the stiffness matrix which is the major core consuming element of the entire routine.

VII. RECOMMENDATIONS

A. ELIMINATE STORAGE LIMITATIONS

The first and most obvious recommendation for further improvement of the routine is to remove the core limitations imposed on the working area. This can be accomplished by the use of direct access devices and a block solving routine which sectionalizes the matrices and works in core with only one section at a time.

It should be noted that the implementation of these modifications will be a difficult task. Substantial coding changes will be required and the difficulties encountered are anticipated to be orders of magnitude greater than those associated with the in-core routine.

B. PARAMETER STUDY

It is normally a part of verifying and testing a FEM program to perform parametric studies on several problems having known exact or approximate results. Time did not permit doing such an extensive investigation with NAT1-3D and it is recommended that appropriate attention be given to a systematic study of the effects of varying the values of NGP, WELD, and the degree of the shape functions and of varying the fineness of the ultimate discretization in each of the three coordinate directions. The literature on FEM describes many such studies and can be used to guide such an

investigation. However, it may be stated that such previous investigations have not led to any significant understanding of the effects of varying parameters in general. Apparently, as of yet, it is important to carry out such an investigation for each new FEM code.

APPENDIX A

PROGRAM DESCRIPTION

A complete dissection and detailed description of the program is not practical. This appendix is intended to contain information which will assist both a reader who desires to pursue a more in-depth study of the problem and a reader who wishes only to make an intelligent casual use of the program.

A. TERMINOLOGY

The following is a listing and brief description of important constant, vector, and matrix names. Functions, definitions and dimensions are given if applicable.

1. Constants

FACT	Value of the Euclidean norm of the mode vector, used to determine iteration convergence
NBAND	Half band width of the stiffness matrix
NEL	Number of elements
NEQ	Number of equations, $NEQ = 3 \times NNODE$
NGP	Number of Gauss points used in the Gaussian quadrature formula
NMAT	Number of materials
NNBC	Number of bounded nodes
NNODE	Total number of nodes
NPEL	Number of nodes per element

2. Vectors (size)

B EMODE FMODE	} (NEQ)	Working vectors used to store the mode shape during iteration and while evaluating the Rayleigh quotient
BGM		Diagonalized mass matrix stored as a vector

3. Matrices (dimensions)

BGK(NEQ,NBAND)	Global or system stiffness matrix
COORD(NNODE,3)	Matrix containing global coordinates of nodes
ENW(3*NPEL,3*NPEL)	Element stiffness or mass matrix
NCONN(NEL,NPEL+1)	Connectivity matrix
NBC(30,4)	Matrix of bounded nodes (i.e., nodes on which boundary conditions are to be imposed) and their boundary conditions

B. PROGRAM ARRANGEMENT

As indicated in Figure 4 the routine is of modular construction. This format allows for maximum utilization of existing software and facilitates future revisions.

1. MAIN

The first segments of the MAIN program initiate the calculation of the large system matrices. Data is passed to the various subroutines via common statements. The iterative portion of MAIN is based upon the decomposition of BGK and the repeated solution of the matrix equation $[BGK] \{v\} = \omega^2 [BGM] \{v\}$. Once convergence has been attained MAIN controls the calculation of the Rayleigh quotient which serves as both a check and a refinement of the frequency obtained by the iteration.

ARRANGEMENT OF NAT1-3D

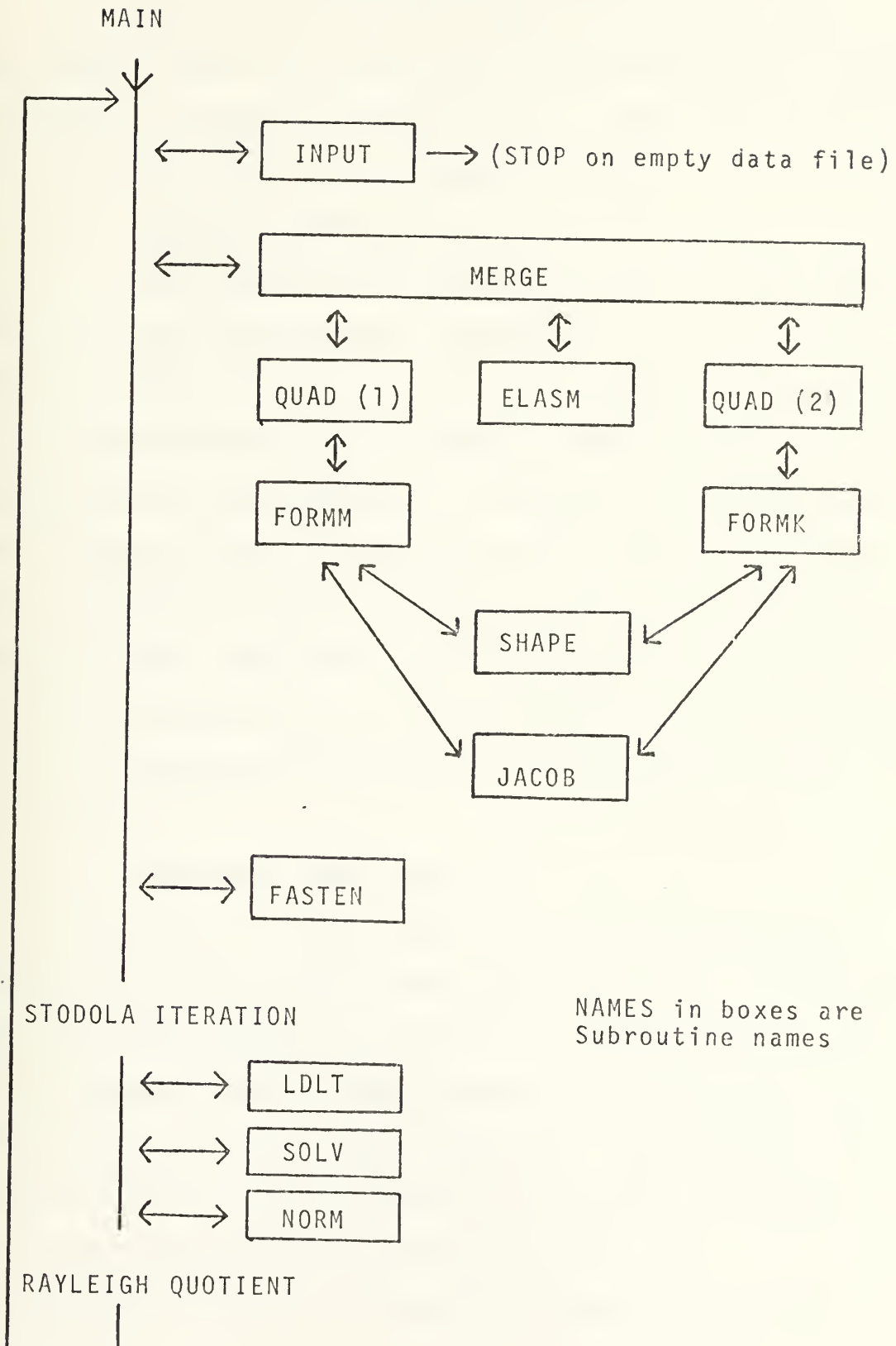


Figure 4

2. Subroutine INPUT

INPUT reads and echo checks the problem data. The most common source of errors in a large routine is a simple typographical error in the data deck. INPUT also computes the half band width of the system matrices.

3. Subroutine MERGE [Ref. 4]

MERGE combines the contributions of each element and distributes them to the corresponding global matrix, [BGK] or [BGM]. The distribution allocation is controlled by the connectivity matrix (NCONN). MERGE serves in a sequence controlling capacity throughout the calculation of the system stiffness and mass matrices. BGM is computed first and during its assembly is stored in the region set aside for BGK. Calculation of the weight of the object provides a quick check on the mass matrix by giving the user an evaluation which he may compare to the actual weight of the object.

4. Subroutine QUAD [Ref. 4]

QUAD is divided into two basic sections which differ in the form of the shape functions used. Computing the mass matrix requires evaluation of the shape functions while computing the stiffness matrix requires evaluation of the derivatives of the shape functions. QUAD selects the weighting factors and integration points that correspond to the number of Gauss points desired by the user. It then calls the appropriate entry of Subroutine FORM which evaluates the

integrand of the Gaussian quadrature formula. Next QUAD multiplies the result by the Gaussian weighting factor and computes the summation. Diagonalization of the mass matrix is also accomplished in QUAD.

5. Subroutine ELASM [Ref. 3]

ELASM forms the constitutive elasticity matrix for each element based on its material which is assumed to be linear, isotropic and elastic.

6. Subroutine FORM [Ref. 4]

FORM evaluates the product of the shape functions or the product of their partial derivatives at each selected Gauss point in the local coordinate system. Dual entry points are used to distinguish between the computations necessary to form the elemental mass matrix and the elemental stiffness matrix.

7. Subroutine SHAPE [Ref. 4]

SHAPE performs the actual evaluation of the shape functions and their derivatives with respect to the local coordinate system. Provisions are included for both linear and quadratic shape functions.

8. Subroutine JACOB [Ref. 4]

JACOB evaluates the Jacobian matrix as well as its inverse and its determinant.

9. Subroutine FASTEN

FASTEN spatially fixes the bounded nodes in the desired coordinate direction by multiplying the appropriate diagonal elements of $[BGK]$ by 10^{10} so as effectively to suppress the associated deflection component. This is a commonly used method, in FEM analysis, of introducing fixity at boundary nodes.

10. Subroutine LDLT [Ref. 5]

LDLT decomposes the system stiffness matrix $[BGK]$ into the products of three matrices: a lower triangular matrix $[L]$ having unit values on the diagonal, a diagonal matrix $[D]$, and $[L]^T$ the transpose of $[L]$; viz. $[BGK] = [L][D][L]^T$. In the process $[BGK]$ is destroyed and replaced by the elements of $[D]$ and the nontrivial elements of $[L]$. The diagonal matrix is stored in the first column and the unit triangular matrix is stored in the remaining space.

11. Subroutine SOLV [Ref. 5]

SOLV performs a forward substitution followed by a back substitution to solve the system $[BGK] \{X\} = \{B\}$. The answers, $\{X\}$, are returned in vector $\{B\}$.

12. Subroutine NORM

NORM calculates the Euclidean norm of the mode vector. This norm is equal to the square root of the sum

of the squares of the element values in the vector. Each element in the vector is then divided by the norm which results in normalizing the vector.

C. CORE ESTIMATION

The mesh generating scheme chosen by the user should inform him of the number of nodes and the bandwidth that have resulted from the discretization. TRIMEG clearly lists this data. With the aid of the following equation these figures may be used to estimate the core requirements which result from the mesh.

$$\text{"K" required} = 110 + \frac{24 \times \text{NNODES} \times (\text{NBAND} + 4)}{1024}$$

The resultant answer includes the necessary buffers and can be used directly as the region specified on the FORTCLG control card.

COMPUTER PROGRAM LISTING

[illegible]


```

000000      DO 43 I=1,NEQ
000000      EMODE(I)=B(I)
000000      CALL NORM
000000      DO 44 I=1,NEQ
000000      FMODE(I)=EMODE(I)
000000      IF(ITER.EQ.I) GO TO 39
000000      DELL = TEST-FACT
000000      IF(OABS(DELL).GT.ACC) GO TO 39
000000      WRITE(6,400) ITER
000000      FORMAT( //,T25,' AFTER',I4,' ITERATIONS THE FIRST MODE VECTOR HAS'
000000      1 CONVERGED TO')
000000      WRITE(6,401)(I,EMODE(I),I=1,NEQ)
000000      FORMAT( //,44(/,10(I4,F9.6)))
000000
000000      THE RAYLEIGH CALCULATION
000000
000000      DO 50 I=1,NEQ
000000      FMODE(I)=EMODE(I)
000000      B(I)=EMODE(I)
000000      DO 51 I=1,NEQ
000000      EMODE(I)=ZRO
000000      DENOM = ZRO
000000      DO 52 I=1,NEQ
000000      DENOM = DENOM+FMODE(I)*FMODE(I)*BGK(I)
000000      DO 60 I=1,NEQ
000000      FMODE(I)=BGK(I,I)
000000      DO 61 I=1,NEQ
000000      BGK(I,I)=I.OOO
000000      DO 62 I=1,NEQ
000000      EMODE(I) = ZRO
000000      K=O
000000      DO 62 J=I,NEQ
000000      K=K+1
000000      IF(K.GT.NBAND) GO TO 62
000000      EMODE(I)=EMODE(I)+B(J)*BGK(I,K)
000000      CONTINUE
000000      TOP = O.OOO
000000      DO 63 I=1,NEQ
000000      TOP = TOP+FMODE(I)*EMODE(I)
000000      WRITE(6,600) TOP,DENOM
000000      FORMAT( //,T25,' TOP = ', G25.16,T0X,' DENOM = ',G25.16)
000000      WS = TOP*3.8706D2/DENOM
000000      W = DSQR(T(W))
000000      WRITE(6,601) W
000000      FORMAT( //,T25,' THE VALUE OF OMEGA FROM THE RAYLEIGH QUOTIENT IS'
000000      1,G25.16)
000000      GO TO 1
000000      END

```



```

C      SUBROUTINE INPUT
C      IMPLICIT REAL*8(A-H,O-Z)
C      COMMON /ENTER/COORD(NNODE,3),NCONN(NEL,NPEL+1)
C      COMMON /ELAS/ D(6,6),ELCON(10,3)
C      COMMON /FIX/ NBC(30,4),TITLE(10),NNBC
C      COMMON /SING/NEL,NBAND,NEQ,NPEL,NGP
C      READ PROBLEM PARAMETERS
C      ZRO = 0.0D0
C      READ(5,100,END=9000) TITLE
C      FORMAT(10A8)
C      WRITE(6,101) TITLE
C      FORMAT(//,15X,10A8,/)
C      READ(5,200) NNODE,NEL,NMAT,NNBC,NPEL,NGP
C      FORMAT(6I10)
C      WRITE(6,201) NNODE,NEL,NMAT,NNBC,NPEL,NGP
C      111X, 'THE NUMBER OF NODES IS',10X,110,/,
C      211X, 'THE NUMBER OF ELEMENTS IS',7X,110,/,
C      311X, 'THE NUMBER OF MATERIALS IS',6X,110,/,
C      411X, 'THE NUMBER OF BOUNDED NODES IS',2X,110,/,
C      511X, 'THE NUMBER OF GAUSS POINTS USED IS',18,///)
C      NEQ = NNODE*3
C      ZERO THE DATA MATRICES
C      K = NPEL + 1
C      DO 10 I=1,NEL
C      DO 10 J=1,K
C      NCONN(I,J) = 0
C      DO 20 I=1,NNODE
C      DO 20 J=1,3
C      COORD(I,J) = ZRO
C      READ MATERIAL PROPERTIES
C      DO 30 I=1,NMAT
C      READ(5,300) IMAT,(ELCON(IMAT,J),J=1,3)
C      300 FORMAT(11I0,3F10.2)
C      WRITE(6,301)
C      301 FORMAT(//, ' MATERIAL PROPERTIES',/,3X, ' MAT.NO. YOUNG,S MOD.

```



```

1 ISSON,S RAT. DENSITY',//)
DO 31 I=1,NMAT
31 WRITE(6,302) I,(ELCON(I,J),J=1,3)
302 FORMAT( 6X,I13,1X,4G15.6)

C C C
      READ THE JOINT COORDINATES

401 WRITE (6,401)
      FORMAT(//,I25,'COORDINATES OF THE JOINTS',//,' JOINT NUMBER',8X,
1,X COORDINATE',14X,'Y COORDINATE',11X,'Z COORDINATE',//)
DO 40 I=1,NNODE
      READ(5,400) IJT,(COORD(IJT,J),J=1,3)
400 FORMAT( 6X,I10,3F15.5)
      WRITE(6,402)(IJT,COORD(IJT,J),J=1,3)
402 FORMAT( 5X,I3,12X,6I4.5,2(3X8G14.5))
40 CONTINUE

C C C
      READ ELEMENT CONNECTIVITY
      AND COMPUTE HALF BAND WIDTH

499 WRITE(6,499)
      FORMAT(//,10X,' THE CONNECTIVITY MATRIX',//)
DO 50 I=1,NEL
      READ(5,500)(NCONN(I,J),J=1,K)
500 FORMAT( 10I5,//11I5)
50 WRITE(6,501)(NCONN(I,J),J=1,K)
501 FORMAT(10X,2I15)
      II = NPEL - 1
      NBAND = 0
DO 53 I=1,NEL
DO 51 K=1,NPEL
      NBC(K,1) = NCONN(I,K)
51 DO 52 J=1,II
      JK = J+1
DO 52 K=JK,NPEL
      NBAND = MAX0(NBAND,IABS(NBC(J,1)-NBC(K,1)))
52 CONTINUE
      NBAND = (NBAND+1)*3
      WRITE(6,502) NBAND
502 FORMAT(//,11X,' SYSTEM HALF BAND WIDTH = ',I5,///)

C C C
      READ BOUNDARY CONDITION DATA

DO 60 I=1,NNBC
60 READ(5,601)(NBC(I,J),J=1,4)
601 FORMAT( 15X,4I5)
      WRITE(6,602)

```



```

50  LL = 3*(J-1)+L
    BGK(II,JJ) = BGK(II,JJ) + ENW(KK,LL)*RHO
    CONTINUE
    SUM = ZRO
    DO 60 I=1,NEQ
    60  BGM(I) = BGK(I,1)
        SUM = SUM + BGM(I)
    SUM = SUM/3.0DO
    WRITE(6,65) SUM
    65  FORMAT( //, ' WEIGHT CHECK *** TOTAL WEIGHT = ',G15.6,///)
C
C
    NOW FORM THE STIFFNESS MATRIX
    DO 70 I=1,NEQ
    DO 70 J=1,NBAND
    70  BGK(I,J) = ZRO
    DO 100 IK=1,NEL
    DO 80 NN=1,NPEL
    80  LJT(NN) = NCONN(IK,NN)
        LM(NN) = (NCONN(IK,NN)-1)*3
    DO 90 II=1,NPEL
    I2=LJT(II)
    DO 90 JI=1,3
    90  CORD(II,J1) = CORD(I2,J1)
        L = NPEL + 1
        MATN = NCONN(IK,L)
        E = ELCON(MATN,1)
        PR = ELCON(MATN,2)
        CALL ELASM(E,PR)
        CALL QUAD(2,NPEL,NGP)
    DO 100 I=1,NPEL
    DO 100 J=1,NPEL
    DO 100 K=1,3
    100  II = LM(I) + K
        KK = 3*(I-1) + K
        DO 100 L=1,3
        JJ = LM(J) + L-II+1
        IF(JJ.LE.0) GO TO 100
        LL = 3*(J-1)+L
        BGK(II,JJ) = BGK(II,JJ) + ENW(KK,LL)
    CONTINUE
    RETURN
    END
C
C
    SUBROUTINE ELASM(E,PR)
    IMPLICIT REAL*8(A-H,O-Z)
C

```



```

DO 60 K=1,6
60 ENN(I,J)=ENN(I,J)+B1(I,K)*B(K,J)
DO 80 I=1,NW
DO 80 J=1,NW
80 ENN(I,J)=ENN(I,J)*DTJ
ENN(J,I)=ENN(I,J)
RETURN
END

SUBROUTINE SHAPE (XI,ETA,ZETA,NPEL)
IMPLICIT REAL*8(A-H,O-Z)

COMMON /DRONE/ AJI(3,3),AJ(3,3),DCOL(20,3),CORD(20,3),COL(20),DTJ

DIMENSION CORDX(4),CORDY(4),CORDZ(4)

DATA CORDX/1.0D0,-1.0D0,-1.0D0,1.0D0/
DATA CORDY/1.0D0,1.0D0,-1.0D0,-1.0D0/
DATA CORDZ/1.0D0,-1.0D0,1.0D0,-1.0D0/

FL(XI,ETA,ZETA,X1,Y1,Z1)=(1.0D0+XI*X1)*(1.0D0+ETA*Y1)*(1.0D0+ZETA*
Z1)/8.0D0

DFL(ETA,ZETA,X1,Y1,Z1)=X1*(1.0D0+ETA*Y1)*(1.0D0+ZETA*Z1)/8.0D0

FOC(XI,ETA,ZETA,X1,Y1,Z1)=(1.0D0+XI*X1)*(1.0D0+ETA*Y1)*(1.0D0+ZETA
1*Z1)*(XI*X1+ETA*Y1+ZETA*Z1-2.0D0)/8.0D0

FQM(XI,ETA,ZETA,Y1,Z1)=(1.0D0-XI*X1)*(1.0D0+ETA*Y1)*(1.0D0+ZETA*
1Z1)/4.0D0

DFQC(XI,ETA,ZETA,X1,Y1,Z1)=X1*(1.0D0+ETA*Y1)*(1.0D0+ZETA*Z1)*(2.0
1D0*X1+XI+ETA*Y1+ZETA*Z1-1.0D0)/8.0D0

DFQMX(XI,ETA,ZETA,Y1,Z1)=-X1*(1.0D0+ETA*Y1)*(1.0D0+ZETA*Z1)/2.0D0

DFQMY(XI,ZETA,Y1,Z1)=Y1*(1.0D0-XI*X1)*(1.0D0+ZETA*Z1)/4.0D0

IGO = (NPEL-8)/12 + 1
GO TO (100,200), IGO

      LINEAR SHAPR FUNCTIONS

100 ICOL = 0
DO 10 IZ=1,2

```



```

DO 10 IX=1,4
  ICOL = ICOL + 1
  XI = CORDX(IX)
  YI = CORDY(IX)
  ZI = CORDZ(IZ)
  COL(ICOL) = FL(XI,ETA,ZETA,XI,YI,ZI)
  DCOL(ICOL,1) = DFL(ETA,ZETA,XI,YI,ZI)
  DCOL(ICOL,2) = DFL(ZETA,XI,YI,ZI,XI)
  DCOL(ICOL,3) = DFL(XI,ETA,ZI,XI,YI)
GO TO 40
10
CCCCCCCC
      QUADRATIC SHAPE FUNCTIONS
      CORNER NODES
      1,3,5,7,13,15,17,19
200 CONTINUE
DO 20 IZ=1,2
  II=12*(IZ-1)+1
  IT=II+6
  IX=0
DO 20 ICOL=II,IT,2
  IX=IX+1
  YI=CORDX(IX)
  XI=CORDX(IX)
  ZI=CORDZ(IZ)
  COL(ICOL)=FQC(XI,ETA,ZETA,XI,YI,ZI)
  DCOL(ICOL,1)=DFQC(XI,ETA,ZETA,XI,YI,ZI)
  DCOL(ICOL,2)=DFQC(ETA,ZETA,XI,YI,ZI,XI)
  DCOL(ICOL,3)=DFQC(ZETA,XI,ETA,ZI,XI,YI)
20
CCCCC
      MID SIDE NODES
      2,6,14,18,4,8,16,20, 9,10,11,12
IC9=8
DO 30 IZ=1,2
  II=12*(IZ-1)+2
  IT=II+4
DO 30 IC2=II,IT,4
  IC4=IC2+2
  IC9=IC9+1
  IN=IC9-8
  XI=CORDX(IN)
  YI=CORDY(IN)
  ZI=CORDZ(IN)
  COL(IC2)=FQM(XI,ETA,ZETA,ZI,YI)
  COL(IC4)=FQM(ETA,ZETA,XI,YI,-ZI)

```



```

C
C
COMMON /FIX/ NBC(30,4),TITLE(10),NNBC
WELD=1.0D10
DO 20 I=1,NNBC
DO 10 K=2,4
IFIX = NBC(I,K)
IF(IFIX.EQ.0) GO TO 10
JTFIX = NBC(I,1)*3-3*IFIX
BGK(JTFIX,1) = WELD*BGK(JTFIX,1)
CONTINUE
C
10
20
RETURN
END
C
SUBROUTINE NORM
IMPLICIT REAL*8(A-H,O-Z)
COMMON /ITER/ EMODE(NEQ),FACT
COMMON /SING/ NEL,NBAND,NEQ,NPEL,NGP
SUM = 0.0D0
DO 10 J=1,NEQ
SUM = SUM+EMODE(J)*EMODE(J)
FACT=DSQRT(SUM)
DO 20 K=1,NEQ
EMODE(K)=EMODE(K)/FACT
WS = 3.8706D2/FACT
WRITE(6,100) W
FORMAT(//,10X,' STODOLA OMEGA = ',G25.16)
100
RETURN
END
C
SUBROUTINE LDLT(N,M)
IMPLICIT REAL*8(A-H,O-Z)
COMMON /BIGS/BGK(NEQ,NBAND),ENN(3*NPEL,3*NPEL),ENW(3*NPEL,3*NPEL),BGM(NEQ)
AN = N
ZRO=0.0D0
SUM = ZRO
DO 30 I=1,N
SUM = SUM+BGK(I,1)
ABGN = SUM*1.0D20
APZRO = ABGN*1.0D-30/AN
AVSN = APZRO/ABGN
NM = N-1
DO 300 I=1,NM
DIAG = BGK(I,1)
30
300
END
FAST0060
FAST0070
FAST0080
FAST0090
FAST0100
FAST0110
FAST0120
FAST0130
FAST0140
FAST0150
FAST0160
FAST0170
FAST0180
FAST0190
NORM0010
NORM0020
NORM0030
NORM0050
NORM0060
NORM0070
NORM0080
NORM0090
NORM0100
NORM0110
NORM0120
NORM0130
NORM0140
NORM0150
NORM0160
NORM0170
LDLT0010
LDLT0020
LDLT0030
LDLT0040
LDLT0060
LDLT0070
LDLT0080
LDLT0090
LDLT0100
LDLT0110
LDLT0120
LDLT0130
LDLT0140
LDLT0150
LDLT0160
LDLT0170

```



```

IF(DIAG.EQ.ZRO) GO TO 400
IF(DIAG.LE.APZRO) WRITE(6,2000) I
DO 100 J=2,M
IF(DABS(BGK(I,J)).LE.AVSN) BGK(I,J)=ZRO
100 BGK(I,J) = BGK(I,J)/DIAG
DO 300 J=2,M
L = I+J-1
IF(L.GT.N) GO TO 300
AA = BGK(I,J)*DIAG
IF(AA.EQ.ZRO) GO TO 300
DO 200 K=J,M
ML = I+K-J
200 BGK(L,ML) = BGK(L,ML)-AA*BGK(I,K)
300 CONTINUE
GO TO 5000
400 WRITE(6,1000) I
1000 FORMAT(//,5X,'MATRIX IS SINGULAR',115)
GO TO 5000
2000 FORMAT(//,5X,'WARNING MATRIX IS NEARLY SINGULAR EXECUTION CONTI
      2      INUES',115,5X,' RESIDUAL ITERATION MAY YIELD ERRONEOUS RESULTS
5000 CONTINUE
      2      RETURN
      END
C
SUBROUTINE SOLV(N,M)
IMPLICIT REAL*8(A-H,O-Z)
COMMON /BIGS/BGK(NEQ,NBAND),ENN(3*NPEL,3*NPEL),ENW(3*NPEL,3*NPEL),BGM(NEQ)
COMMON /SOL/B(NEQ)
NM=N-1
DO 100 I=1,NM
BI=B(I)
DO 100 J=2,M
L=I+J-1
IF(L.GT.N) GO TO 100
B(L) = B(L)-BGK(I,J)*BI
100 CONTINUE
DO 200 I=1,N
B(I) = B(I)/BGK(I,1)
DO 300 L=2,N
IR=N-L+1
BIRP=B(IR+1)
DO 300 J=2,M
IRO=IR-J+2

```


SOLV0250
SOLV0260
SOLV0270
SOLV0280
SOLV0290

IF(IRO.LE.0) GO TO 300
B(IRO) = B(IRO) - BGK(IRO,J)*BIRP
300 CONTINUE
RETURN
END

APPENDIX C

MESH GENERATION USING TRIMEG

The purpose of this appendix is to give the user who wishes to use TRIMEG a few brief but very important suggestions that could save time and reduce frustration.

A. BACKGROUND

TRIMEG [Ref. 1] is an automatic mesh generator for use with three dimensional isoparametric elements. It is both quick and efficient and saves the user a lot of tedious, time consuming work by punching the bulk of the data cards required to solve the vibrational problem.

B. COMMENTS

When using TRIMEG there are a few subtle guides which must not be overlooked in order to obtain the most efficient discretization.

1. Orientation

Unless the user has a reasonably good idea of the purposes and functions of TRIMEG it is likely that he will not use NAT1-3D to the greatest advantage. He is likely to orient his super elements or provide for subsequent subdivision of super elements into a number of individual elements in a disadvantageous way so as to use up available storage while accommodating a smaller total number of elements than would have been possible by a more effective arrangement.

Accordingly he is urged to study Ref. 2 before undertaking a problem. In particular, referring to Figure 13, p. 29, Ref. 2, the following general remarks may be of assistance.

The super elements all have the same orientation given by ξ , η , and ζ directions. A super element is divided into ROW parts in the η direction, into COL parts in the ξ direction, and into SLICE parts in the ζ direction. The body represented by the super elements has, let us say, a maximum of n_{ξ} super elements in the ξ direction, a maximum of n_{η} super elements in the η direction, and a maximum of n_{ζ} super elements in the ζ direction. The orientation of the super elements should be chosen so that the numerical products (ROW)(n_{ξ}), (COL)(n_{η}), and (SLICE)(n_{ζ}) are in ascending order. This gives minimum bandwidth corresponding to the total number of elements into which the body is finally subdivided.

2. Boundary Nodes

For configurations having curved boundaries it is possible and advisable to use 32 boundary nodes to describe the super element even when linear or quadratic shape functions are to be used. The use of the additional nodes allows TRIMEG to more accurately follow the curvature of the boundary of the object as it subdivides the super element.

C. LISTING OF DATA DECK FOR TRIMEG

```

*****
** MESH GENERATING PROGRAM FOR TRISOP,
** WITH QUADRATIC ELEMENTS
** CODED BY J.R. ADAMEK, MARCH 1973, NAVAL POST GRADUATE SCHOOL
**
*****

```

THE NECESSARY INPUT FOR THIS PROGRAM IS AS FOLLOWS:

CARD 1 AND 2, TITLE, FORMAT (6A8) EACH CARD.

COL. 1 TO 48 BRIEF TITLE OF PROBLEM. USER'S NAME AND/OR BOX NO.
MUST BE INCLUDED ON ONE OF THE CARDS IF CALCOMP
OPTION INCLUDED. (SEE CARD 3)

CARD 3, MESH PARAMETERS, FORMAT (3I5,3F5.0)

COL. 1 TO 5 (NSEL): NUMBER OF SUPER ELEMENTS.

6 TO 10 (NPUNCH): IF THIS IS ZERO OR BLANK, NO CARDS WILL
BE PUNCHED. IF NPUNCH IS DIFFERENT FROM ZERO A DECK
FOR CONNECTIVITY AND X, Y & Z COORDINATES OF JOINTS
WILL BE PUNCHED. NPUNCH BLANK OR ZERO SHOULD ALWAYS
BE USED UNTIL ONE IS SATISFIED WITH THE MESH.

11 TO 15 (NPLOT): IF THIS IS ZERO OR BLANK, NO PLOTS WILL
BE MADE. IF NPLOT IS (1) A PLOT WILL BE OBTAINED ON
THE CALCOMP PLOTTER.
ALL PLOTS WILL BE PLOTTED IN FIRST QUADRANT ONLY.

THE FOLLOWING THREE INPUT VARIABLES ARE THE EULER ANGLES.
NOT REQUIRED UNLESS NPLOT = 1.

16 TO 20 (TETA): ANGLE OF ROTATION ABOUT Z-AXIS.

21 TO 25 (ALPHA): ANGLE OF ROTATION ABOUT LATEST X-AXIS.

26 TO 30 (BETA): ANGLE OF ROTATION ABOUT LATEST Y-AXIS.

SUPER ELEMENT DECK, TOTAL OF NSEL CARDS, FORMAT (14I5), LOAD IN
ASCENDING ORDER OF SUPER ELEMENT NUMBERS.

COL. 1 TO 5 (SEL NO.) SUPER ELEMENT IDENTIFICATION NUMBER.

6 TO 10 (ROW): NUMBER OF ROWS IN SUPER ELEMENT.

11 TO 15 (COL): NUMBER OF COLUMNS IN SUPER ELEMENT.


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16 TO 20 (SLICE): NUMBER OF SLICES IN SUPER ELEMENT.
*****
21 TO 25 (NODE A) SUPER ELEMENT
26 TO 30 (NODE B) CONNECTIVITY
31 TO 35 (NODE C) COUNTERCLOCKWISE
                      (SEE NOTE)
36 TO 40 (NODE D) *
41 TO 45 (NODE E) A *****
46 TO 50 (NODE F) * * * * *
51 TO 55 (NODE G) * * * * *
56 TO 60 (NODE H) * * * * *
*****

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NOTE: NUMBERING OF SUPER-ELEMENT CORNER NODES IS ARBITRARY. HOWEVER, CORNER NODES WHICH ARE CONNECTED TO OTHER CORNER NODES, MUST HAVE THE SAME NUMBER.

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61 TO 65 (TYPE): MATERIAL IDENTIFICATION NUMBER FOR
SUPER ELEMENT.(SEE 'TRISCP' INST)
66 TO 70 (NPTSB): NUMBER OF SUPER ELEMENT BOUNDARY NODES.
NPTSB = 8,20 OR 32

```

SUPER ELEMENT BOUNDARY DECK, TOTAL OF NPTB CARDS, (315,3F20.0).

NPTB=SUM OF BOUNDARY NODES OF ALL SUPER ELEMENTS. ORDER IS COUNTER CLOCK-WISE ABOUT ZETA AXIS, STARTING AT NODE COMMON WITH (XI,ETA,ZETA)=(1,1,1). IF MORE THAN ONE SUPER ELEMENT, LOAD ITS BOUNDARY AFTER THAT OF FIRST SUPER ELEMENT IN SAME NODAL ORDER AS ABOVE. ORDER OF SUPER ELEMENTS SAME AS FOR SUPER ELEMENT DECK.

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COL. 1 TO 5 (SEL NO.): SUPER ELEMENT IDENTIFICATION NUMBER.
6 TO 10 (NODE NO.): ARBITRARY NUMBERING OF SUPER ELEMENT
BOUNDARY NODES. MAY OR MAY NOT BE THE

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11 TO 15      (NCT): TYPE OF COORDINATES. IF THIS FIELD IS
                BLANK OR ZERO, CARTESIAN COORDINATES ARE LOADED
                IN FORMAT DESCRIBED BELOW. NCT=(1) FOR MID SIDE
                NODE OF QUADRATIC SUPER ELEMENTS WITH PARTICULAR
                SIDE LINEAR. NCT=(2) FOR MID SIDE NODES OF CUBIC
                SUPER ELEMENTS WITH PARTICULAR SIDE LINEAR. LOAD
                ZERO OR BLANK FOR X, Y AND Z COORDINATES WHEN NCT
                IS (1) OR (2).

16 TO 35      X COORDINATE OF NODE, REAL*8

36 TO 55      Y COORDINATE OF NODE, REAL*8

56 TO 75      Z COORDINATE OF NODE, REAL*8

ADDITIONAL PROBLEMS MAY ALSO BE LOADED AFTER FIRST PROBLEM.

STORAGE REQUIREMENTS ARE AS FOLLOWS:
160K FOR NO PLOT; 180K FOR CALCOMP.
REPRESENTATIVE EXAMPLES USING AROUND EIGHT SUPER ELEMENTS
EXECUTED IN APPROXIMATELY 1 MINUTE AND 10 SECONDS.

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11 TO 15
(NCT): OR ZERO, CARTESIAN COORDINATES ARE LOADED
IN FORMAT DESCRIBED BELOW. NCT=(1) FOR MID SIDE
NODE OF QUADRATIC SUPER ELEMENTS WITH PARTICULAR
SIDE LINEAR. NCT=(2) FOR MID SIDE NODES OF CUBIC
SUPER ELEMENTS WITH PARTICULAR SIDE LINEAR. LOAD
ZERO OR BLANK FOR X, Y AND Z COORDINATES WHEN NCT
IS (1) OR (2).

```

16 TO 35	X COORDINATE OF NODE,	REAL*8
36 TO 55	Y COORDINATE OF NODE,	REAL*8
56 TO 75	Z COORDINATE OF NODE,	REAL*8

ADDITIONAL PROBLEMS MAY ALSO BE LOADED AFTER FIRST PROBLEM.

STORAGE REQUIREMENTS ARE AS FOLLOWS:
160K FOR NO PLOT; 180K FOR CALCOMP.
REPRESENTATIVE EXAMPLES USING AROUND EIGHT SUPER ELEMENTS
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